NASA-CR-204369

N93-25
9N-37-CR
324142

LIQUID PHASE TRANSIENT BEHAVIOR OF A HIGH TEMPERATURE HEAT PIPE

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INTRODUCTION

Transporting thermal energy at a high rate over a small temperature gradient is an important require-ment for many heat transfer applications. The heat pipe provides a passive heat transfer system that accomplishes this requirement.

There have been many advances in heat pipe design, theory and practice since the first experimental work performed by Grover (1964) and the first quantitative analysis performed by Cotter (1965). The scope of published information spans a broad range of theory, analysis, experiment, and application. However, the majority of these works are limited to steady state applications under highly idealized conditions. Comparatively little work has been done that addresses future heat transport requirements for applications that would benefit from the use of heat pipes. For example, space missions requiring high power supplied and reliable operation over a wide range of conditions may use nuclear generating systems. Such a system has a wide range of heat transfer requirements that may be met by heat pipes.

The purpose of this research is to develop an analytical model of the transient dynamics of the liquid phase in a high temperature heat pipe. This liquid phase model is intended to be coupled with a vapor phase model for the assessment of the potential for heat pipes to meet future heat transfer requirements covering a broad spectrum of transient operating conditions. These requirements are assumed to include high temperatures, high thermal flux, and reliable operation over long distances and small temperature drops. The model is intended to calculate the temperatures, velocities, and pressures required to assess heat pipe transients. Transient operation characteristics of interest include the capability of a heat pipe to reach steady state operation following a change in operating conditions; the allowable magnitudes and rates of changes; the time required to respond to transient conditions; and the temperatures, velocities, and pressures encountered during transient operation.

Intensive studies have been carried out at the Los Alamos National Laboratory on startup and shut-down operations of heat pipes (Merrigan 1985 and 1986, and Merrigan et al. 1986). Experiments were performed on long heat pipes using liquid sodium and potassium as working fluids. Bystrov and Goncharov (1983) performed experimental and theoretical studies on a heat pipe with and without a foreign gas. In their analysis they used a lumped parameter approach for heat balance. Chang and Colwel (1984 and 1985) and Jang (1988) also studies the startup transient of heat pipes. Jang (1988) performed an overall analysis of the liquid and vapor phases. However, the vapor core was assumed at steady state.

The heat pipe selected for this study uses an annular wick configuration as shown in Figure 1. The wick configuration consists of several layers of fine pore screen pressed together and concentrically installed in the pipe to form an annular flow channel. The annulus forms a low resistance flow channel while the fine pore screen tube forms the high pumping capability boundary between the liquid and vapor regions.

THE SOLUTION PROCEDURE

The objective of this study is to develop an analytical model of the liquid phase of a heat pipe. The model is intended to be coupled to a vapor phase model for the complete solution of the heat pipe problem. The mathematical equations are formulated consistent with physical processes while allowing a computationally efficient solution. The model simulates time dependent characteristics of concern to the liquid phase including input and output heat fluxes, phase change, liquid temperatures, container temperatures, liquid velocities, and liquid pressures.

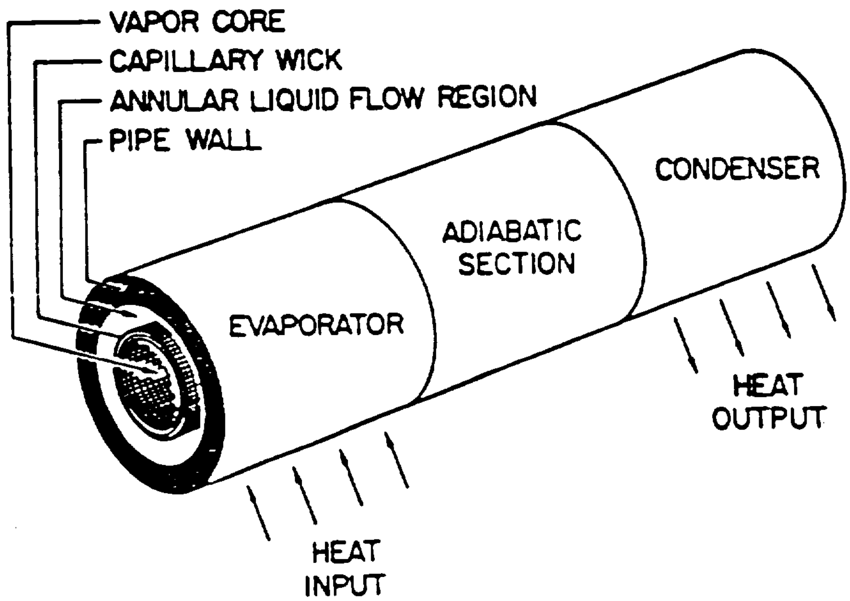

FIGURE 1. Heat Pipe with High Performance Composite Annular Wick.

The heat pipe selected for study, see Figure 1, has an annular wick consisting of several layers of fine pore screen concentrically installed in the pipe forming an annular flow channel. The screen tube vapor side capillary forces yield the pumping capability that drives the flow in the annulus.

The governing equations for the liquid are conservation of mass momentum and energy with appropriate boundary conditions. The system is assumed to operate in a gravity free environment. The thermal energy source adds heat uniformly around the evaporator circumference while the heat sink extracts heat uniformly around the condenser circumference. The liquid flow in the annulus is relatively simple because the cross-stream dimension is much smaller than the streamwise allowing the cross-stream pressure gradient to be ignored. The boundary conditions on the flow are that the velocities are zero on the bounding surfaces and the ends are adiabatic.

The heat pipe external boundary is divided into three regions. The evaporator region absorbs energy from the environment, the adiabatic region exchanges no energy with the environment and the condenser rejects heat to the environment. The evaporator surface is assumed to be exposed to a known surface heat flux condition, while the condenser exchanges heat radiatively to a sink of known temperature.

The liquid vapor interface serves as the communication link between the liquid and the vapor along the entire axial length of the pipe. The general liquid-vapor interface conditions are conservation of mass, continuity of tangential stress, continuity of normal stress, continuity of thermal flux, and supplemental relations deriving from kinetic theory.

A very simplified treatment of the vapor phase was implemented to close the mass flow circuit. It was felt that the vapor response being orders of magnitude faster than the liquid response, it could be treated as quasi-steady. Below the sonic limit the vapor space is considered to be a single control volume. The mass in the volume is found by keeping tract of the evaporation and condensation. With the vapor space volume an estimate of the density can be made in terms of the liquid-vapor interface and saturation temperatures. With the Claussius-Clapeyron equation the density can be found. This is used with an equation of state to yield an equation for the liquid-vapor interface temperature in terms of the vapor saturation temperature which closes the problem. If the sonic limit is reached, two control volumes are postulated that communicate mass at the sonic limit. Keeping tract of the mass in each and assuming that the limit is reached at the exit of the evaporator allows a treatment similar to the subsonic case to be carried out.

The governing equations, boundary conditions, property relations and supplemental equations were coded for solution in FORTRAN and computations were carried out using the UCLA IBM 3090. The detailed algorithms and code listing are given by Roche (1988).

DISCUSSION OF RESULTS

Capabilities of the computational algorithm were tested by running two heat pipe startup cases. The heat pipe dimensions were chosen to be consistent with the experimental device described by Merrigan et al (1986) up to the limit of the uniform grid used in this analysis. The experimental heat pipe in that study used lithium as the working fluid and an annular wick configuration. The annular region for liquid flow was formed between the pipe interior wall and a porous concentric tube constructed of 7.25 layers of pressed screen. The high temperatures involved required the container to be constructed from the refractory allow molybdenum. The heat pipe geometry parameters used in this analytical study compared to the actual parameters are

Parameter	Actual	Approximate
Heat Pipe Internal Length (L)	4.0	4.0 m
Evaporator Length (L_e)	0.4 m	0.4 m
Condenser Length (L_c)	3.0 m	3.0 m
External Diameter (D_w)	1.90 cm	1.886 cm
Internal Diameter (D_l)	1.60 cm	1.598 cm
Vapor Space Diameter (D_y)	1.49 cm	1.490 cm
Effective Wick Pore Diameter (D_p)	53 μm	53 μm

where the effective wick pore diameter was experimentally determined from surface tension measurements.

An axial grid step size of $h_x = 0.2$ m and a radial grid step size of $h_r = 1.1 \times 10^{-4}$ m were used. The computational grid consisted of 21 axial grid steps and 20 radial grid steps. The grid mapping to the heat pipe configuration is:

Radial Liquid Space	$1 \le i < 6$
Radial Pipe Wall	$6 \le j \le 20$
Axial Evaporator Region	$1 \le i \le 3$
Axial Adiabatic Region	$1 < i < 6$
Axial Condenser Region	$6 \ge i \le 21$

The time step size was selected by running a series of calculations for three hundred time steps each with gradually decreasing step sizes. A time step size of $\delta_1 = 1 \times 10^{-3}$ second was found to eliminate inconsistent liquid-vapor interface temperatures, but did not completely remove pressure oscillations. A time step size of $\delta_1 = 1 \times 10^{-3}$ second removed the pressure oscillations for the three hundred time steps calculation. This time step was used in the calculations.

SLOW STARTUP TEST

A startup test was devised to supply input energy to the evaporator such that the heat pipe would not reach the sonic limit. This test was to check overall program implementation. Input heat flux was supplied according to the function

$$q_{in} = \frac{1}{2} q_{max} \left\{ 1 - \sin \left[\pi \left(\frac{t}{\tau} + \frac{1}{2} \right) \right] \right\} \quad t \le \tau$$

$$q_{in} = q_{max} \qquad t > \tau$$

where q_{max} is the maximum input heat flux and τ is the time stretching parameter that controls how quickly q_{in} reaches q_{max}. In this test, $q_{max} = 1 \times 10^6$ W/m^2 and $\tau =$ 300 seconds corresponding to a five minute period for the input heat flux to reach the maximum.

Calculations were performed for 113,193 time steps, or a period slightly greater than 114 seconds. The run was terminated at this time since the liquid pressure drop exceeded the capillary pumping limit. Analysis of the results shows that the capillary pumping limit was exceeded due to a large jump in pressure drop at a single time step. The pressure calculations had been experiencing oscillations with usually small amplitudes of approximately 2.5 Pa. The oscillations began after the 300 steps used to verify the adequacy of the time step size of 1×10^{-3} second. The oscillations become very large at $t = 90$ seconds, but begin to decay until the large jump occurs at approximately 114 seconds.

Power transfer as a function of time is shown in Figure 2. While the input power briefly exceeds the sonic limit line, the phase change heat transfer remains below the sonic limit. At approximately 103 seconds into the transient, the sonic limit exceeds the capillary limit so that the capillary limit becomes the controlling heat transport limit. At approximately 105 seconds, the unlikely result is produced that the phase change heat transfer exceeds the input heat transfer. With the exception of this anomaly, the heat transfer curves are well behaved, smooth functions of time. In addition, the phase change heat transfer is well below the capillary limit heat transfer at the point when liquid pressure drop jumps to exceed the capillary pumping limit. The cause of the liquid pressure drop jump is not apparent from the heat transfer curves.

The vapor temperature as a function of time is shown in Figure 3. The vapor temperature is also a smooth, although rapidly increasing, function of time. The cause of the jump in liquid pressure drop is also not apparent from vapor temperature.

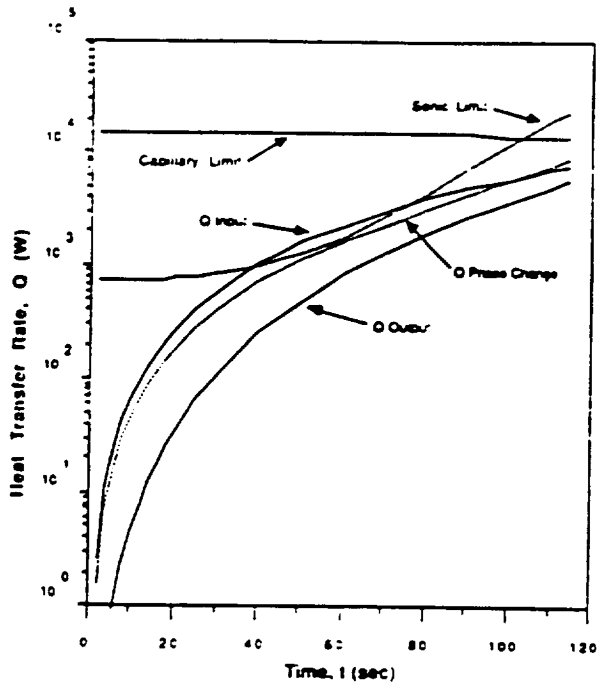

FIGURE 2. Slow Startup Heat Transfer Results.

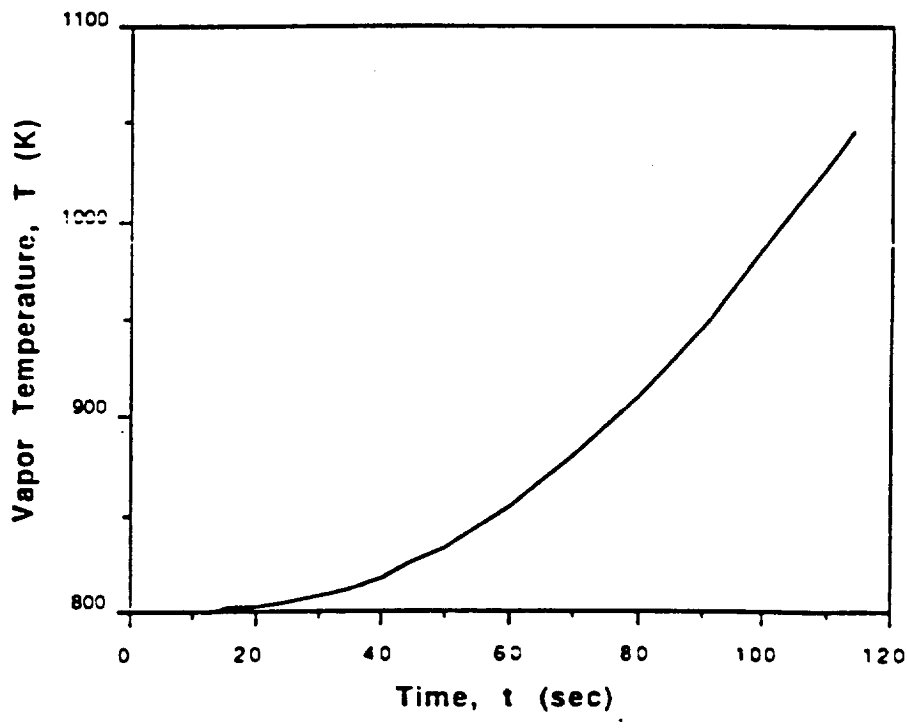

FIGURE 3. Slow Startup Vapor Temperature Results.

There are many potential causes for the pressure oscillations. The pressure oscillations may be physical, but this possibility is remote considering the numerical sensitivities of the method. The time step size of 1 x 10^{-3} second may have been too large. If this is the case, a smaller time step may not be acceptable considering the cost of computations for a transient process that occurs over a period of tens of minutes.

The pressure calculations are based on the convenient assumption that the minimum liquid pressure occurs at the end of the condenser, although this assumption is not generally valid for the wide range of operating conditions encountered during a startup transient.

SONIC LIMIT STARTUP TEST

A startup test was performed to test algorithm calculation of sonic limited operation. Using again $\tau = 300$ seconds, the maximum input heat flux was increased to $q_{max} = 1.5 \times 20^6$ W/m^2 to assure the phase change heat transfer would reach the sonic limit. Heat transfer results are shown in Figure 4.

Input power exceeds the sonic limit at 20 seconds, and remains above the sonic limit through the duration of the test of 130 seconds. The evaporation and condensation heat fluxes are essentially equal as the sonic limit is reached at approximately 40 seconds. Evaporation and condensation follow the sonic limit curve until 80 seconds. After 80 seconds, condensation heat flux increases above the sonic limit, while evaporation heat flux continues to follow the sonic limit curve. This unlikely result is a result of the inadequacies of the simple vapor model used to perform calculations. The vapor model unintentionally forces a high condensation rate, which forces the condenser temperature to increase rapidly due to the high thermal resistance of the radiation boundary condition on the condenser. This temperature effect is shown in Figure 5. The evaporator vapor temperature smoothly and gradually increases with time, while the

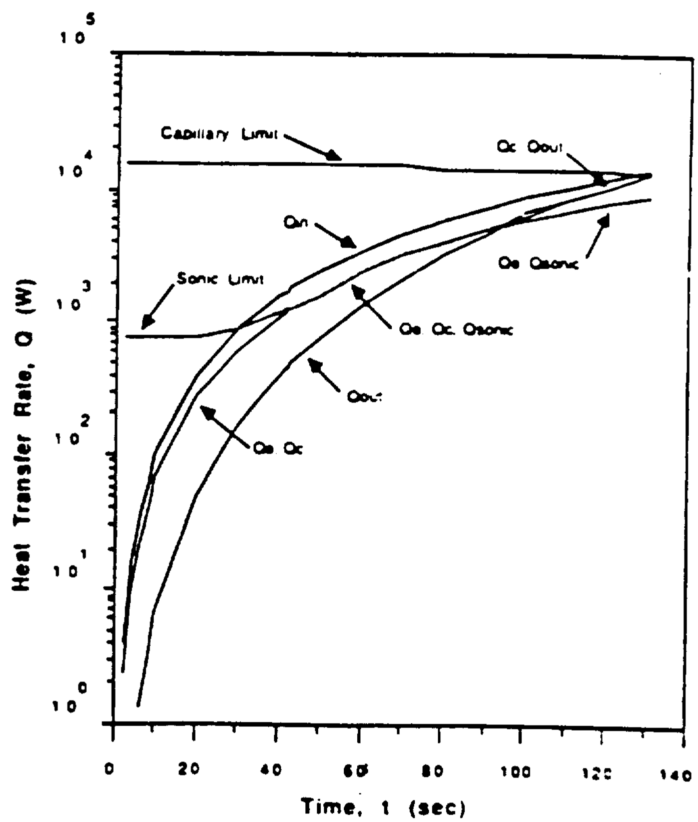

FIGURE 4. Sonic Limit Startup Heat Transfer Results.

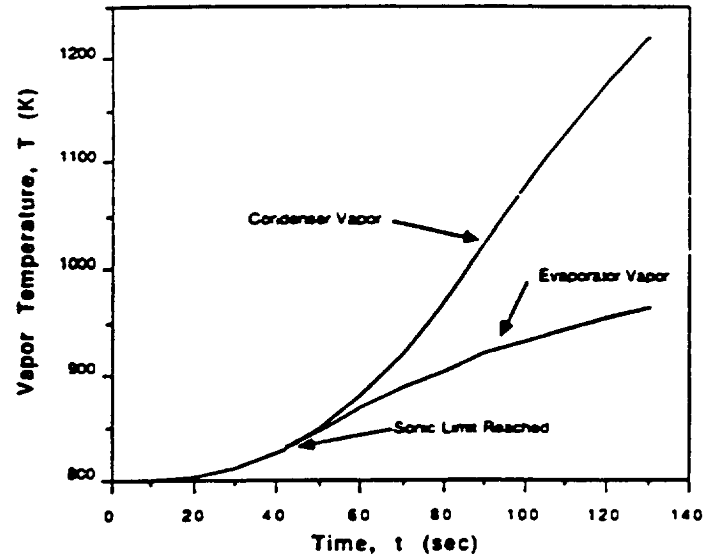

FIGURE 5. Sonic Limit Startup Vapor Temperature Results.

condenser vapor temperature rapidly increases. The condenser temperature eventually increases sufficiently so that radiation heat transfer from the condenser becomes very efficient. At approximately 93 seconds, the output heat transfer exceeds the evaporation heat transfer, which is another unlikely result. Calculations are terminated at 130 seconds due to the condensation heat transfer exceeding the capillary heat transfer limit. While the capillary limit is not strictly a limit on condensation. Calculations had already passed beyond physical significance and the computational implementation had been demonstrated.

CONCLUSIONS AND RECOMMENDATIONS

The heat pipe is a very complex device such that performing analysis of heat pipe transients is a difficult venture. The difficulty is due to the coupled liquid and vapor dynamics, the phase change process, the coupled heat transfer problem with nonlinear boundary conditions, the physical geometry, and the transient time requirement. This study has presented the full system of governing equations, including boundary conditions, required to solve the liquid phase heat pipe problem. A simplified solution was formulated by using certain assumptions and integrating the liquid dynamics equations.

The resulting formulation identified a key requirement for future research in solving the heat pipe problem. This study used a kinetic theory approach to model the phase change. The model consists of a large coefficient multiplying a very small temperature difference. The model requires a very small time step for stability even with the simplifications used in this analysis. The large coefficient also transforms temperature differences that are otherwise beyond machine accuracy into computationally significant terms. Future research should investigate an alternative model for the phase change process.

Difficulty with geometry was encountered due to the widely disparate length scales in the different coordinate directions. The radial dimension across the liquid is much smaller than the radial dimension across the pipe wall, which in turn is much smaller than dimensions in the axial direction. One approach to address this problem is to assume radial gradients are negligible. The radial pressure gradient was neglected in this study. This study also found small radial temperature gradients indicating that the radial thermal resistance is small. The drawback of this approach is the loss of fidelity with the true physics of the problem. A preferable approach may be to use a variable mesh grid with a to-be-determined simplified treatment of radial gradients. The variable mesh grid allows fine resolution in important regions with coarse resolution in the other regions. The simplified treatment of radial gradients would take advantage of the small radial gradients.

The long duration of transients presents a competing concern with model complexity. An overly complex model with high fidelity may be too computationally expensive for practical calculations of transients.

This study has also shown that the transient heat pipe problem is not amenable to a straightforward simplification such as is used for the vapor phase. The solution of the heat pipe transient problem requires a full solution of the liquid and vapor phases. This elusive solution is left to future efforts.

Acknowledgment

This work was supported by Nasa Lewis under Contract No. NAG3-899 and NASA Dryden under Contract No. NCC2-374 Supp2.

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